

PAPER • OPEN ACCESS

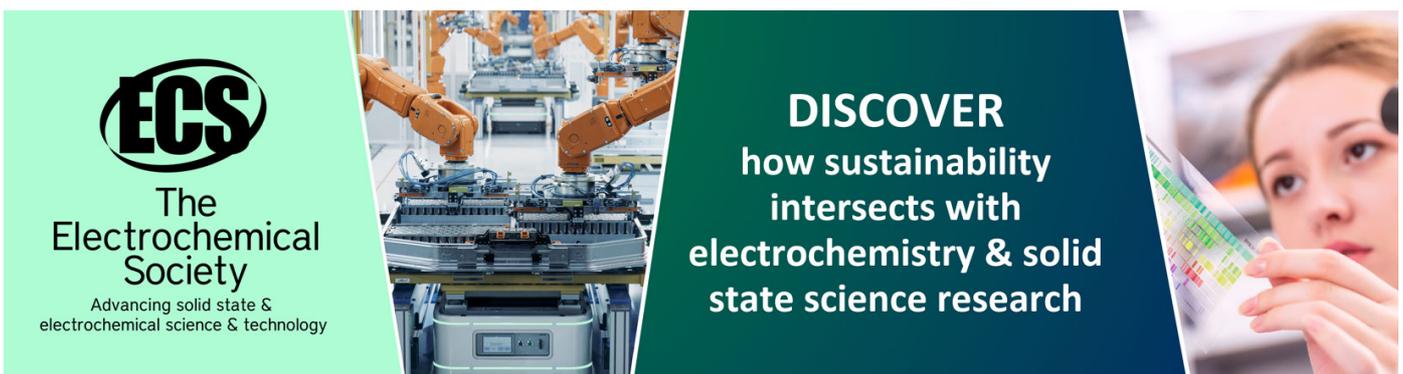
Cost of enlarged operating zone for an existing Francis runner

To cite this article: Christine Monette *et al* 2016 *IOP Conf. Ser.: Earth Environ. Sci.* **49** 072018

View the [article online](#) for updates and enhancements.

You may also like

- [An engineering approach for the prediction of fatigue damage in the part load regime](#)
F Von Locquenghien and B Leibing
- [Validation of deep part load dynamic stresses for axial runners](#)
C Monette, J Chamberland-Lauzon and B Nennemann
- [Francis turbine dynamic stresses measurements at model and prototype scales](#)
Chamberland-Lauzon J., DiSciullo J., Monette C. et al.



ECS
The
Electrochemical
Society
Advancing solid state &
electrochemical science & technology

DISCOVER
how sustainability
intersects with
electrochemistry & solid
state science research

Cost of enlarged operating zone for an existing Francis runner

Christine Monette¹, Hugues Marmont¹, Joël Chamberland-Lauzon¹, Anders Skagerstrand², André Coutu¹, Jens Carlevi³

¹Andritz Hydro Canada Inc., 6100 Trans Canada Hwy., Pointe-Claire, H9R 1B9, Québec, Canada

²Vattenfall Vattenkraft AB, Norra Torget 4, 681 30 Kristinehamn, Sweden

³Vattenfall AB, R&D Laboratories, 814 26 Älvkarleby, Sweden

christine.monette@andritz.com

Abstract. Traditionally, hydro power plants have been operated close to best efficiency point, the more stable operating condition for which they have been designed. However, because of changes in the electricity market, many hydro power plants operators wish to operate their machines differently to fulfil those new market needs. New operating conditions can include whole range operation, many start/stops, extensive low load operation, synchronous condenser mode and power/frequency regulation. Many of these new operating conditions may impose more severe fatigue damage than the traditional base load operation close to best efficiency point. Under these conditions, the fatigue life of the runner may be significantly reduced and repair or replacement cost might occur sooner than expected.

In order to design reliable Francis runners for those new challenging operating scenarios, Andritz Hydro has developed various proprietary tools and design rules. These are used within Andritz Hydro to design mechanically robust Francis runners for the operating scenarios fulfilling customer's specifications. To estimate residual life under different operating scenarios of an existing runner designed years ago for best efficiency base load operation, Andritz Hydro's design rules and tools would necessarily lead to conservative results.

While the geometry of a new runner can be modified to fulfil all conservative mechanical design rules, the predicted fatigue life of an existing runner under off-design operating conditions may appear rather short because of the conservative safety factor included in the calculations. The most precise and reliable way to calculate residual life of an existing runner under different operating scenarios is to perform a strain gauge measurement campaign on the runner. This paper presents the runner strain gauge measurement campaign of a mid-head Francis turbine over all the operating conditions available during the test, the analysis of the measurement signals and the runner residual life assessment under different operating scenarios. With these results, the maintenance cost of the change in operating mode can then be calculated and foreseen by the power plant owner.

1. Introduction

The last 20 years of development in the Swedish electricity market has brought in more wind, less nuclear, new free market solutions, increased transfer capacity to other countries, environmental issues (EU water frame directive) and much lower price on electricity. Some of these are consequences of political decisions. The installed capacity of wind has increased from almost zero to about 15% (2014) of electricity production capacity and about 7-8 % of the energy.



In parallel it has been found that the stability with respect to, for example, frequency in the grid has degraded. This is supposed mainly to be caused by the technical changes of the system. More wind power and less nuclear but also the increased transfer capacity means less inertia and more stochastic electricity production. Wind power is recognized as non-predictable on timescales about one hour or longer and requires flexibility in other parts of the system to compensate variations. The same can be supposed with solar. It is also a risk that wind and solar production can be varying opposite to consumption patterns which further increase the need of flexible production.

The new market solutions also influence and give more complexity to the system and a reduction of the stability has been seen. The development has led to increased need of capacity to compensate for variations in production from wind, which is a need in hour(s) to week(s) perspective and to improve the frequency regulation capability on shorter timescales.

All these changes are supposed to continue, further reduction of nuclear, increased wind, and solar could also be expected to be installed in the Nordic countries. Due to low prices there are no economic incentives to invest in new regulating capacity in the foreseeable future. There are a number of initiatives and research going on to prepare for this new situation and the solution can be found in different ways both on the supply and the demand side and also in the design of the market system but the most important factor, all conclude, is the regulating capacity of hydro.

For the single power plant and unit the consequence is to operate on a much more irregular pattern. Historically the variation has been mainly on a year base between summer and winter and on a day to night base. In a situation with much wind power production the total production with hydro in the system is reduced compared with today and thus the need to do regulating work is drastically increased for the unit. Stornorrfor is a power station near the sea at the end of a river system and by that the use of the turbines for regulation is important as they take care of the water coming from the reservoirs located in the mountain some 400 km to the west, starting a couple of days before, as well as an unregulated river merging just upstream the station. The power station consists of 4 units, G1 and G2 150 MW, G3 131 MW and G4 182 MW and the production is about 2,3 TWh a normal year. Stornorrfor is the hydro power station owned by Vattenfall with the largest production.

Traditionally the operating zones and “forbidden ranges” have been defined through the measurement of pulsations in the water passages. By doing more investigations on the units, like measurements on the runner, it could help to define critical zones for the equipment on a more detailed level, to give input for changes of operation strategies and also to increase the operating range without increasing the mechanical loads over design values. Another aspect is the possibility to understand ageing processes of the machines which is one factor in how to value the new operating patterns needed in the future.

Several papers have been published on the cost of operation of various operating conditions based on runner strain gauge measurements for different specific speeds [1-8]. However, hydro power plant owners could wonder how to evaluate the fatigue life potential of their own specific machines. This paper presents specifically the recent measurement campaign done on Stornorrfor G1 Francis runner and how measurement results are used to evaluate runner fatigue damage under various operating condition and patterns.

2. Measurement campaign

2.1. Turbine description

The strain gauge runner measurement campaign took place from November 24 to December 15, 2015 at the Stornorrfor underground generating station located near Umeå, northern Sweden. G1 unit is a vertical Francis turbine, equipped with a 15 blade runner having a diameter of 4896 mm and replaced in 2005. After 10 years of operation, the runner was found in very good condition. The original powerhouse and turbine originate from 1958. The sectional view of the machine is shown in Figure 1. Stornorrfor G1 runner can generate up to 150 MW for a maximum head of 72.5 m at a rotational speed of 125 rpm.

2.2. Experimental setup

2.2.1. Runner strain gauges. Preparation of this measurement campaign was a key factor in order to optimize the instrumentation process at site. The number of strain gauges and their locations on the runner were defined based on stress results of Finite Element Analysis (FEA). Uniaxial strain gauges were installed close to hot spot stress regions and oriented along with the maximum principal stress direction.

Two successive blades were instrumented with 5 strain gauges per blade for a total of 10 strain gauges. As shown in Figure 1, strain gauges were installed at the blade-to-band and blade-to-crown trailing edge corners on pressure and suction sides. One strain gauge was installed at the blade root to crown on pressure side. Two strain gauges were placed in sequence on one blade at the blade outflow junction with crown on suction side in order to better determine the stress gradient.

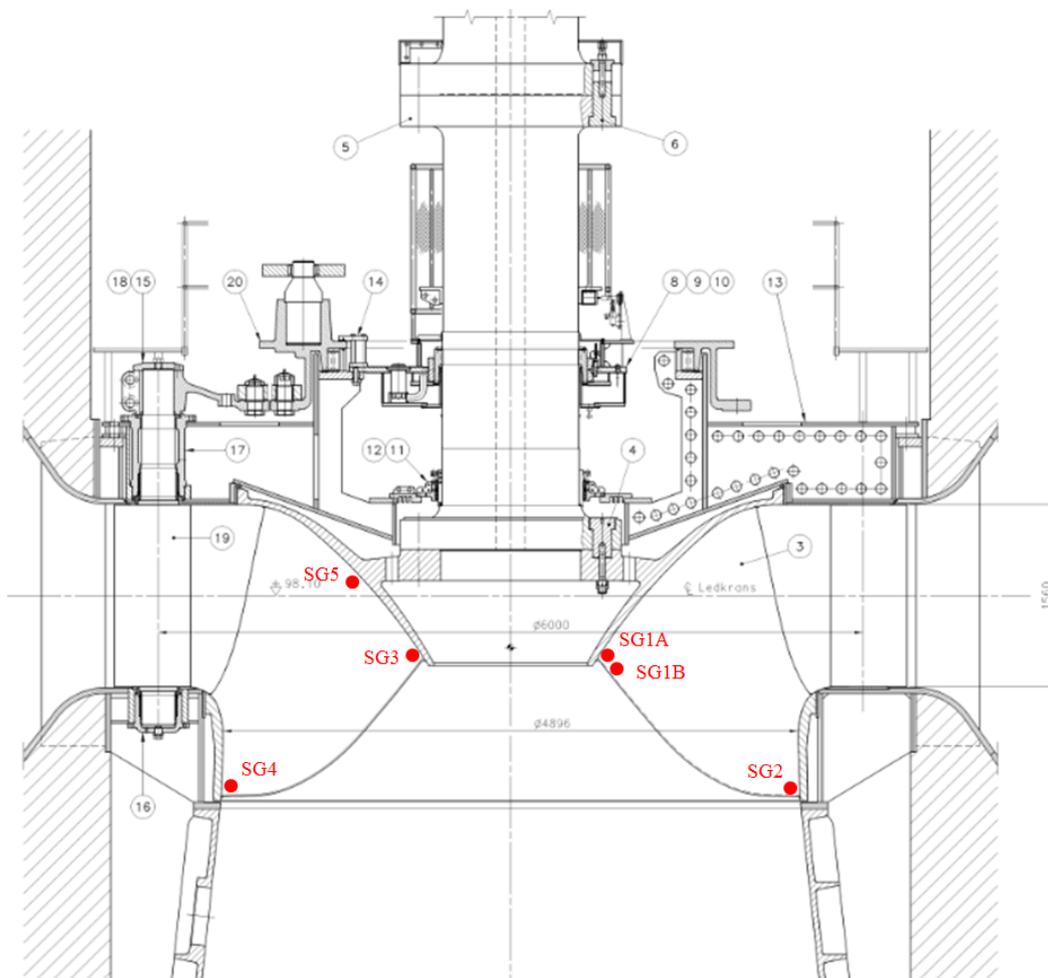


Figure 1. Stornorrfors G1 turbine sectional view and strain gauge locations

The type of strain gauge was carefully considered in order to minimize machine downtime at site. Strain gauge positioning was optimized by means of templates to ensure accurate location of the strain gauges on the runner blade surface. Strain gauges and wires were protected from the high water flow and turbulent flow at some partial load conditions with a wear resistant epoxy resin.

The strain gauge wires were passed inside the runner crown and through the turbine and generator shaft air admission centre bores and were connected to a rotating data acquisition system installed at the top of the generator.

2.2.2. Acquisition system setup. Andritz Hydro used both a rotating data acquisition system for the runner strain gauges, designed to rotate with the generator shaft of the machine, and a stationary data acquisition system for all the stationary instrumentation installed on the fixed part of the machine. Both data acquisition systems were synchronized and data could be monitored in situ.

2.2.3. Stationary instrumentation. Stationary instrumentation was connected to the fixed data acquisition system to record all the operating conditions of the tests. Both Vattenfall Vattenkraft AB and Andritz Hydro collaborated to setup the stationary instrumentation and both parties shared their acquisition of all sensor signals for hydraulic and rotor conditions such as pressures, bearing proximeters, accelerometers, runner speed, power, shaft torque, water levels, rotor up-thrust, air admission flow and guide vane angle.

2.3. Testing procedure

The test campaign consisted of several steady state conditions on a large power range of the machine from speed-no-load to high power. Those steady state conditions were repeated with and without air admission. The usual start-up sequence of the Stornorrfor's unit was also tested. The intent was to possibly optimise the start-up sequence in terms of runner stress measurements and start-up duration.

3. Stress measurements

3.1. Static stresses

Static stresses were predicted by finite element analysis for tested operating conditions. Usual Andritz Hydro steady state CFD design procedure was used to calculate the pressures applied into the flow channel [9]. Variable pressures on the seal surfaces and in the chambers are calculated and applied on the model using analytical formulations. In addition to pressure loads, gravity and centrifugal loads are also applied. The agreement between the static strain prediction at the strain gauge locations and actual mean strain measurements are compared for the two extreme operating conditions of speed-no-load and 136 MW in Figures 2 and 3. The quality of the agreements validates the static stress predictions done using Andritz Hydro procedures.

3.2. Dynamic stresses

Strain signal of a strain gauge 1A, close to the blade outflow junction to the crown, is shown in Figure 4 during a start-up procedure and different power levels from SNL to high power. As it was shown from other runner measurements in the past [1-7], it can be observed that the dynamic stress level is relatively high at speed-no-load and at low power while it gradually decreases with power increase. The frequency content of the signal at each power is shown through a waterfall diagram in Figure 5. From this waterfall diagram, several phenomena can be identified such as the once per revolution hydraulic unbalances $f_n = 2.08$ Hz and its harmonics present at all powers, the part load rope at $0.8 f_n \approx 1.66$ Hz between 90 MW and 125 MW, and the Rotor-Stator Interaction (RSI) phenomenon $20f_n = 41.67$ Hz. RSI phenomenon has received a lot of attention in the literature in the last decade and can be predicted with reasonable accuracy [10-12]. However, for a low head machine such as Stornorrfor's G1, stress amplitudes caused by this phenomenon are expected to be quite low and not to represent a significant contributor to the runner fatigue life. This is confirmed by the measurements done at Stornorrfor's G1 shown in Figure 5. The stochastic stress amplitudes at low and part load conditions, spread from almost 0 to hundreds of Hz, represent the real hurdles for the fatigue life of this runner. Also, the high amount of energy in the strain measurements around 25 to 30 Hz is typical of Francis runner low part load operation and is related to unstable flows such as interblade vortices.

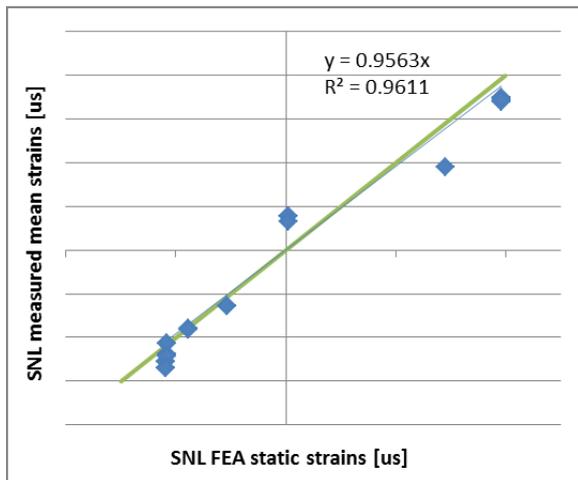


Figure 2. SNL static strain correlation between FEA and measurements at strain gauge locations

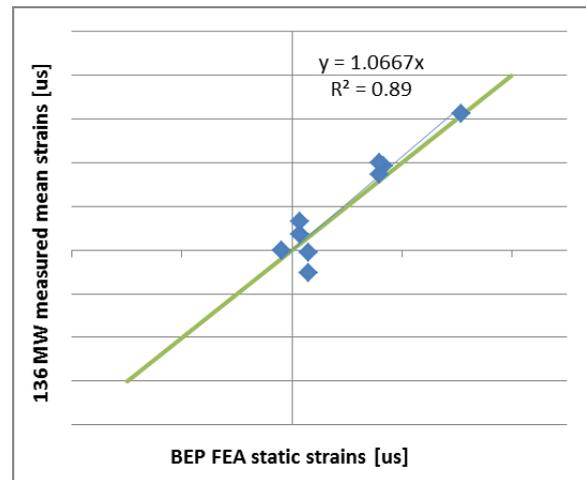


Figure 3. High power static strain correlation between FEA and measurements at strain gauge locations

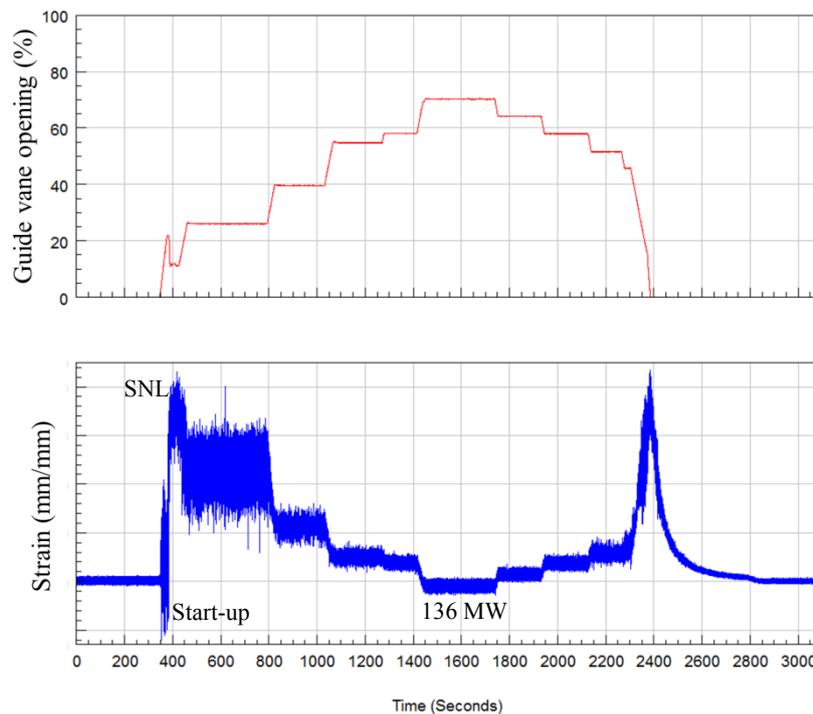


Figure 4. Time signal of strain gauge 1A close to the blade outflow junction to the crown compared to the guide vane opening

Even if recently developed unsteady CFD calculations have shown good agreement with stochastic stress measurements at speed-no-load conditions for several runners [13], the stochastic nature of the dynamic stress at low and part load operating condition makes it necessary to use both extrapolation methods and statistics for its prediction. For new runner designs, Andritz Hydro established conservative statistical criteria to predict the low and part load stochastic stress amplitudes, allowing the design of reliable new runners [2]. For existing runners, strain gauge measurements, such as the ones performed on Stornorrfor G1, allow a very accurate calculation of the runner fatigue cumulated damage at each operating condition.

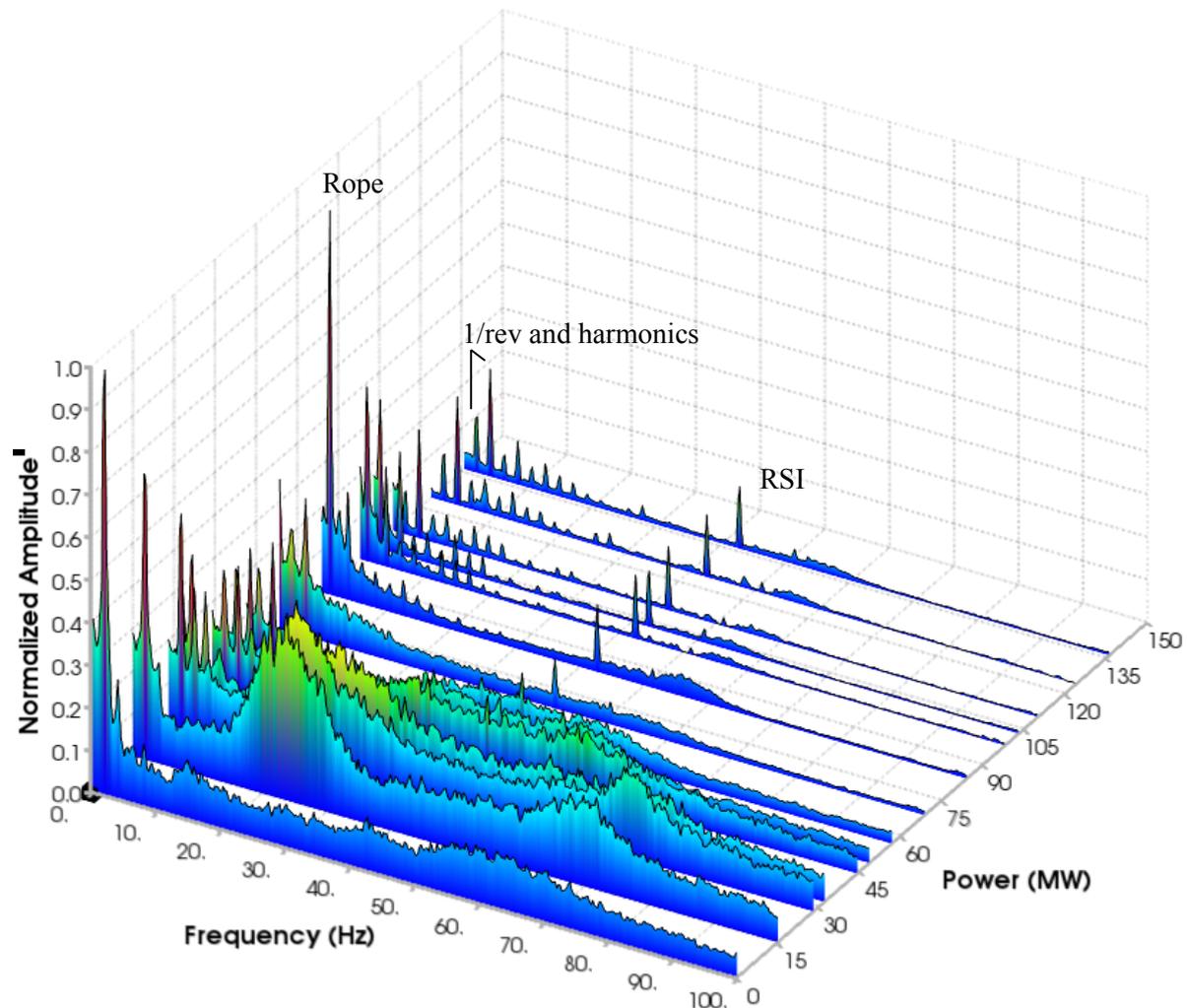


Figure 5. Waterfall normalized amplitude spectrum of the strain gauge 1A close to the blade outflow junction to the crown

3.3. Start-up stresses

Due to very smooth start-up, Stornorrfor's G1 start-up transient maximum dynamic stresses are similar in amplitude to the low part load dynamic stresses as it can be observed on Figure 4. During the measurement campaign, optimisation was deemed unnecessary in regard of runner stresses and it was decided not to modify the implemented start-up sequence.

4. Cost of operation calculation from measurements

4.1. Stress transposition to hot spot

The first step in order to calculate fatigue damage done by measured stresses is to transpose strain measurements made at the strain gauge locations to structure hot spots. For static stresses, maximum stress predictions can directly be used because of the excellent agreement of correlations between measurements and predictions (Figures 2 and 3). For dynamic stresses, only the predicted dynamic stress pattern is considered. Figure 6 shows the dynamic stress pattern calculated by FEA and Figure 7 shows the correlation between the predicted dynamic strain pattern and the maximum measured dynamic strain amplitudes at SNL. Based on the good quality of the correlation, transposition factors

from measured strains to runner hot spots for Stornorrfor's G1 are extracted from the FEA dynamic stress pattern.

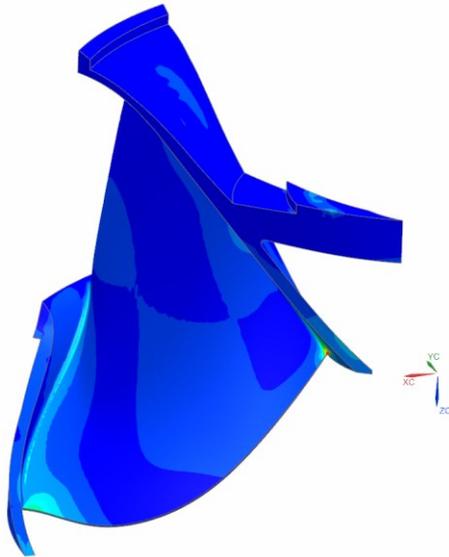


Figure 6. FEA von Mises dynamic stress pattern

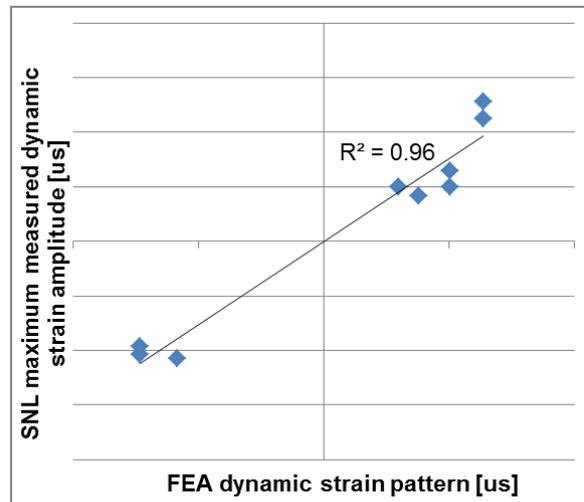


Figure 7. SNL maximum dynamic strain correlation between FEA dynamic strain pattern and measurements at strain gauge locations

4.2. *Rainflow calculation*

The broad band frequency content of the low and part load stochastic stress requires the use of rainflow counting method for the evaluation of fatigue damage. Rainflow counting method makes it possible to include in the fatigue analysis all loading phenomena occurring on the runner. Rainflows of strain gauge measured signals transposed to runner hot spots are shown at a common scale in Figures 8 to 11 for signal samples of 100 seconds at 4 different operating conditions.

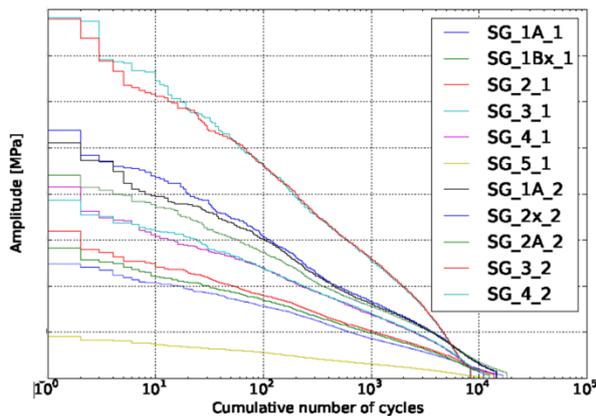


Figure 8. Speed-no-load rainflow of all strain gauge measurements after transposition to blade dynamic hot spots

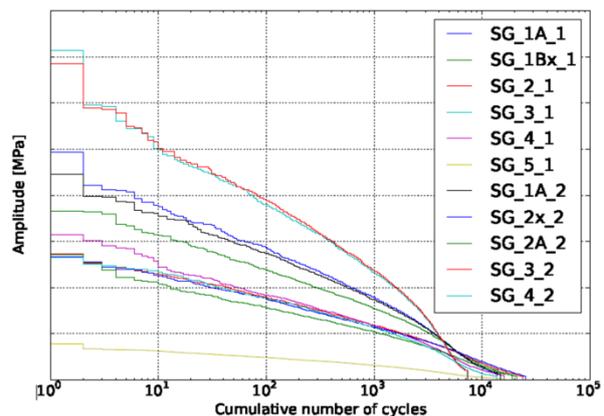


Figure 9. 44 MW rainflow of all strain gauge measurements after transposition to blade dynamic hot spots

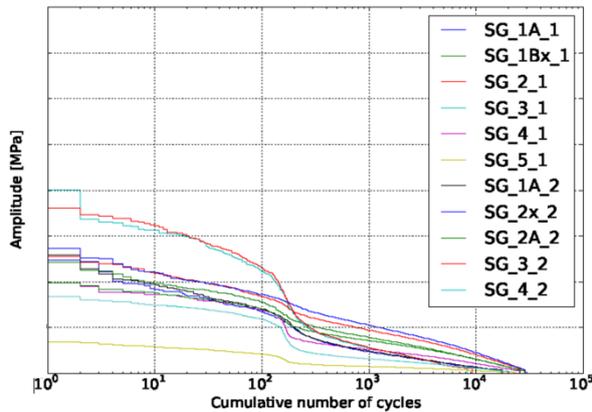


Figure 10. 84 MW rainflow of all strain gauge measurements after transposition to blade dynamic hot spots

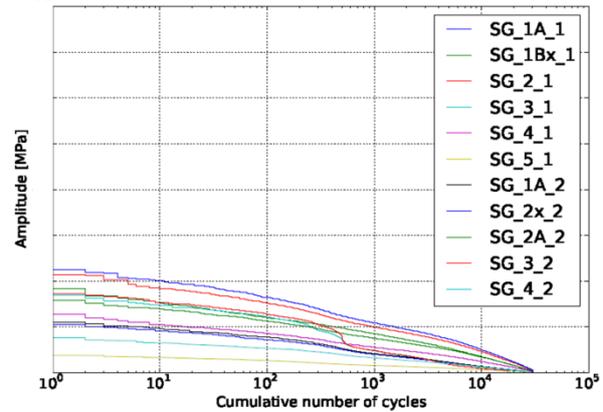


Figure 11. 136 MW rainflow of all strain gauge measurements after transposition to blade dynamic hot spots

At speed-no-load and low load conditions, when the dynamic stress is mainly stochastic, the rainflow shape is mainly triangular with logarithmic axis. At 84 MW, the rope appears as a protuberance with higher amplitude at low number of cycles (~100 cycles). The RSI phenomenon amplitude is too low to be distinguished on the measurement rainflows.

4.3. Life usage calculation

With the use of a design fatigue curve in water, the cumulative Miner’s rule [14] is used to calculate the fatigue life usage of each measured operating condition. The relative cumulated fatigue damage is given for all measured operating conditions in Figure 12 and conservative operating zones are defined with resulting relative cumulated fatigue damage given in Table 1. Start-up transient dynamic stresses were also studied with the rainflow counting method and the relative cumulated damage of a single start-up, compared to 100 seconds of steady state operating conditions, is also indicated in Table 1. The blade outflow to crown junction is the runner location that is the most dynamically solicited and the most sensitive location for possible fatigue failure.

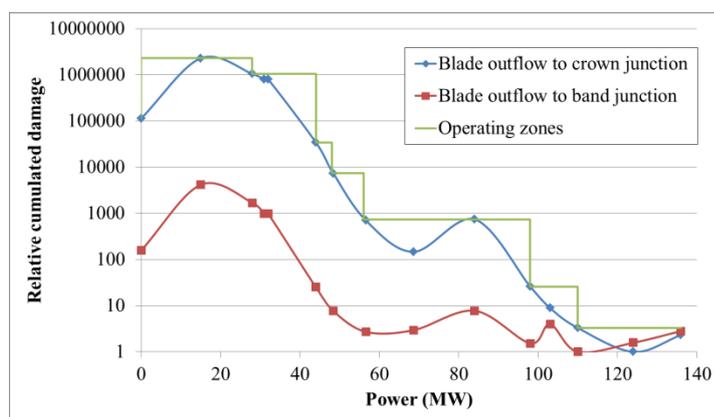


Figure 12. Relative cumulated fatigue damage of the measured operating conditions at Stornorrfor G1

Table 1. Fatigue damage	
Operating zone for 100 sec. or transient event	Relative cumulated fatigue damage
Speed-no-load	114 227
0 to 28 MW	2 293 798
28 to 44 MW	1 050 641
44 to 48 MW	34 373
48 to 56 MW	7 340
56 to 98 MW	741
98 to 110 MW	26
110 to 136 MW	3.3
Start-up from 0 rpm to speed-no-load	390 527

To avoid premature cracking of the runner while using low and part load operation, it is very important to adequately define the past and future operation through Design Reference Mission (DRM) [2-3]. Table 2 shows, through 4 different DRMs, the variability of potential runner fatigue life according to the chosen operation of the turbine. Any operation below 110 MW exponentially decreases the runner fatigue life. Compared to the traditional DRM, the SNL DRM with 10% of the time at speed-no-load is 13.4 times more damaging and would reduce to 7.5% the fatigue life of the runner. Because of the relative smooth start-up, the start-stop DRM with 3 starts-up per day is less damaging than the SNL DRM.

Table 2. Relative fatigue life of 4 different Design Reference Missions (DRM)

Operating zone or transient event	Traditional DRM	SNL DRM	Multiple loads DRM	Start-stop DRM
Speed-no-load	10 min / day	2.4 h / day	2.4 h / day	10 min / day
0 to 28 MW	--	--	2.4 h / day	--
28 to 44 MW	--	--	2.4 h / day	--
44 to 48 MW	--	--	2.4 h / day	--
48 to 56 MW	--	--	--	--
56 to 98 MW	--	--	2.4 h / day	--
98 to 110 MW	--	--	--	--
110 to 136 MW	23.83 h / day	21.4 h / day	12 h / day	23.83 h / day
Start-up	1 / week	1 / week	1 / week	3 / day
Relative cumulated damage	1.0	13.4	400.0	2.5
Relative fatigue life	1.0000	0.0749	0.0025	0.4000

5. Conclusions

Because electricity market is changing with the development of solar and wind energy, grid stability decreases. To improve stability, the regulating capacity of hydro can be exploited which however implies non-negligible equipment maintenance and replacement costs. With the objective of improving our understanding of the hydroelectric turbine ageing process and to optimize future operation patterns, strain gauge measurement campaign was performed on the Stornorrfor G1 runner. Measurement results are processed and analysed. Relative cumulated fatigue damages are calculated for start-up and various steady state operating conditions using FEA stress prediction pattern for transposition to runner hot spots, rainfall counting method, Miner's cumulated damage rule and design fatigue curve in water. Relative fatigue lives of four different DRMs are compared to demonstrate how the choice of operation pattern is deterministic for the runner fatigue life.

All this paper has been written using relative fatigue damage values. Of course, the real damage depends on the stress values and the fatigue curve used. In the evaluation of absolute fatigue life time, a judicious choice of fatigue curve is essential to define meaningful inspection plan in line with chosen operation.

References

- [1] Coutu A., Monette, C., Nennemann B., Chamberland-Lauzon J., Ruchonnet N., Taruffi A., "Specific speed effect on Francis runner reliability under various operating conditions", *Sixth International Conference on Water Resources and Renewable Energy Development in Asia*, Vientiane, Lao PDR, March 2016.
- [2] Coutu A., Marier S., Chamberland-Lauzon J., Monette C., "Designing Francis runners for 0-100 per cent operation", *Hydro 2015*, Bordeaux, France, October 2015

- [3] Coutu A., Chamberland-Lauzon J., “The impact of flexible operation on Francis runners”, *The International Journal on Hydropower & Dams*, Volume Twenty Two, Issue 2, 2015.
- [4] Huang X., Chamberland-Lauzon J., Oram C., Klopfer A., Ruchonnet N., “Fatigue analyses of the prototype Francis runners based on site measurements and simulations”, *27th IAHR Symposium on Hydraulic Machinery and Systems*, Montreal, Canada, 23-25 September 2014
- [5] Coutu A., Chamberland-Lauzon J., Monette C., Nennemann B. 2013 Life consumption of Francis runners under various operating conditions, *5th International Workshop on Cavitation and Dynamic Problems in Hydraulic Machinery*, Lausanne, Switzerland
- [6] Coutu A. 2013 Mechanical Issues Related to Wide Operating Range, CEATI Hydropower Workshop, Mitigating Impacts of Aging Infrastructure, Las Vegas, NV.
- [7] Sick M., Oram C., Braun O., Nennemann B., Coutu A., “HPP delivering regulating power: Technical challenges and cost of operation”, *Hydro 2013*, Innsbruck, Austria, October 2013
- [8] Coutu A., Gagnon M., Monette C. 2007, Life Assessment of Francis Runners Using Strain Gage Site Measurements, *Waterpower XV*, Chattanooga, TN
- [9] Melot M., Monette C., Coutu A., Nennemann B., “A new Francis runner design procedure to predict static stresses at speed-no-load operation”, *The International Journal on Hydropower & Dams*, Volume Twenty One, Issue 1, 2014
- [10] Monette C., Nennemann B., Seeley C., Coutu A., Marmont H., “Hydro-dynamic damping in flowing water”, *27th IAHR Symposium on Hydraulic Machinery and Systems*, Montreal, Canada, 23-25 September 2014.
- [11] Coutu A., Roy M. D., Monette C., Nennemann B., “Experience with Rotor-Stator Interactions in High Head Francis Runner”, *IAHR 24th Symposium on Hydraulic Machinery and Systems*, Foz do Iguassu, Brazil, 2008.
- [12] Coutu A., Monette C., Velagandula O., “Francis Runner Dynamic Stress Calculations”, *Hydro 2007*, Granada, Spain, October 15-17, 2007
- [13] Nennemann B., Morissette J.-F., Chamberland-Lauzon J., Monette C., Braun O., Melot M., Coutu A., Nicolle J., Giroux A.-M., “Challenges in Dynamic Pressure and Stress Predictions at No-Load Operation in Hydraulic Turbines”, *IAHR 27th Symposium on Hydraulic Machinery and Systems*, Montreal, Canada, September 2014.
- [14] Stephens R.I., Fatemi A., Stephens R.R., Fuchs H.O., “Metal Fatigue in Engineering, Second Edition”, John Wiley & Sons, 2001